HEAT TRANSFER TO A PLATE BEHIND AN OBSTACLE

M. G. Ktalkherman

Zhurnal Prikladnoi Mekhaniki Tekhnicheskoi Fiziki, Vol. 7, No. 5, pp. 130–133, 1966

In engineering it is often necessary to deal with stalled flows, such as occur behind flame stabilizers, behind various obstacles, after a sudden change in the cross section of a duct and so on. Great attention has recently been given to the investigation of heat transfer in such flows. This research has been largely experimental owing to the complexity of the theoretical analysis. For instance, in Seban's work [1] heat transfer to a plate in the turbulent flow behind a step was investigated. Charwat's experiments [2] were devoted to heat transfer in a rectangular cavity. This paper deals with heat transfer to a plate behind an obstacle.

The experiments were carried out on the apparatus which was used previously to investigate the effectiveness of jet protection of surfaces [3]. A steel test panel formed the bottom of a rectangular channel of width B = 155 mm and height H = 215 mm. The length of the working part of the test panel was about 1 m and its thickness was 10 mm.

A ten-section plate-type electric heater mounted underneath the panel allowed independent regulation of the power in each section, where the heat flux distribution over the area could be regarded as uniform. Transverse asbestos-filled grooves in the panel prevented heat flow along the surface, while heat insulation reduced the leakage of heat downwards and to the side.

The wall temperature t_w was measured by thirty nichrome-constantan thermocouples embedded in the center of the panel. The air temperature t_0 in the channel was measured by a nichrome-constantan thermocouple held in a textolite fork. The emfs of the thermocouples were measured by a P2/1 potentiometer. The air flow G_0 was determined by integrating the velocity profile measured in the last section of the test panel (to construct the velocity profile we measured the velocity at not less than 40 points in the section). The velocity was measured with a Prandtl tube 2 mm in diameter. The velocity head was measured with a MMN micromanometer (Class 0.5). The power released in the heater sections was determined from the readings of an ammeter and voltmeter (Class 0.5).

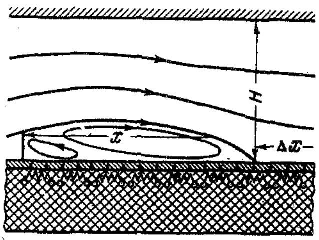


Fig. 1

In calculating the heat flux from the wall to the air we introduced corrections for heat transfer by radiation and heat loss through the insulation. The first correction was introduced by calculation and for the second we carried out special calibration experiments.

The wall temperature in most cases was 40–65° C, and the air temperature in the channel was 20–25° C. The velocity of the air flow $w_1 = G_0/F\gamma_0$ varied in the experiments from 6 to 28 m/sec (here F = HB is the area of the cross section of the channel and γ_0 is the density of air).

In these conditions the total correction for heat loss could be as much as 25% of the measured value. The heat flux over the length was kept constant to within ±10%.

The first series of experiments were of a calibratory nature, during which the local heat transfer on the plate was investigated. The velocity profile in the initial cross section of the working part of the channel was fairly uniform and the boundary layer was turbulent.

The experiments covered the range of Reynolds numbers R_x from $3.5 \cdot 10^4$ to $2 \cdot 10^6$. The experimental points with a scatter of ±10% were predicted by the formula

$$N_x = 0.0263 R_x^{0.8} \qquad (1)$$

which is valid [4] for a smooth plate.

Another similar control experiment was carried out after the end of the main experiments. Its results were also predicted satisfactorily by the above relationship. In the experiments of the main series a flat plate 1 mm thick and as long as the width of the channel was fitted to the test panel at the entrance to the working section of the channel. The angle between the plate and the panel was 90°.

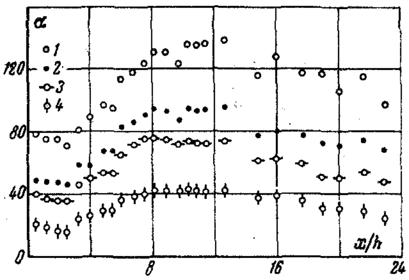


Fig. 2

A schematic illustration of the flow behind the plate is shown in Fig. 1. A region of separation and a region of reattachment can be distinguished.

Abbot et al. [5] and Bearman [6] in experimental investigations of the structure of the flow behind a rectangular step observed two vortices in the region of separation: there was a principal one and another, smaller one, directly adjacent to the step. In the considered case the flow picture would presumably be similar. The experiments were conducted with plates of three sizes with height h equal to 22, 42, and 60 mm. This enabled us to determine the heat transfer at distances of 1 to 50 plate heights including the separated region and the reattached region. In the treatment of the experimental data we took the heat transfer coefficient in every case as the quotient of the specific heat flux transferred by convection and the difference in wall and air temperatures at the entrance: $\alpha = q/(t_w - t_0)$. Figure 2 shows the heat transfer coefficient α (W/m² · deg) plotted against the length x/h in experiments with a plate of height h = 42 mm. Points 1, 2, 3, and 4 are for values of w_1 = 27, 15.6, 11.9, and 5.8 m/sec. Similar

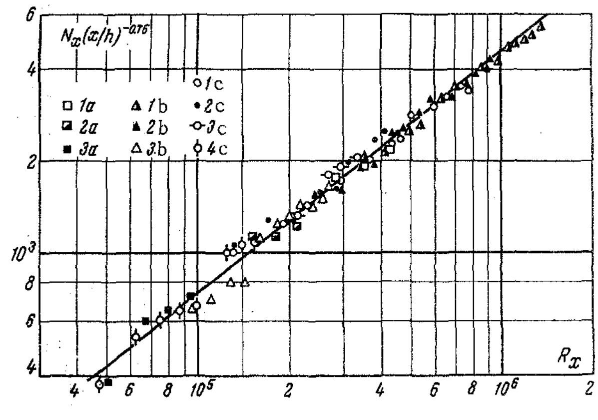
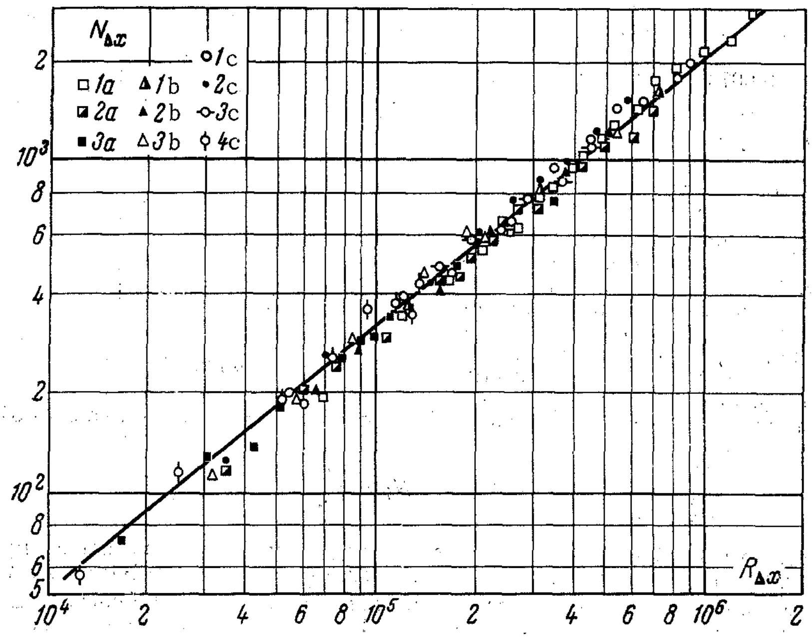


Fig. 3



$N_{\Delta x}$
1a 1b 1c
2a 2b 2c
3a 3b 3c 4c
$R_{\Delta x}$
10^4 2 4 6 8 10^5 2 4 6 8 10^6 2
5 6 8 10^2 2 4 6 8 10^3 2

Fig. 4

relationships were obtained in experiments with plates of heights h = = 22 mm and h = 42 mm, the only difference being that at h = 60 mm the region of reverse flow was better represented and at h = 22 mm the region behind the reattachment point was better represented.

Figure 2 clearly shows the similarity of the curves. They all have a fairly flat maximum occupying a region of 9–12 plate heights and a gently descending right branch. The left branch has a more complex structure: the heat transfer coefficient decreases rapidly at first with increasing distance from the maximum and then, after reaching a minimum (at a distance of approximately 2 plate heights from the plate), increases slightly. The heat transfer coefficient attains its maximum value at the point of reattachment of the flow. However, owing to the flatness of the maximum of the curves and the inaccuracy of the experiment the reattachment point could not be fixed exactly. All we can say is that it lay in the region of 9–12 plate heights.

To find formulas predicting the experimental relationships we divided the whole curve into three regions of the following size: the first from 2 to 9, the second from 9 to 12, and the third from 12 to 50 plate heights.

Measurement of the heat transfer in the first region covered the range of Reynolds numbers R_x from $5 \cdot 10^4$ to $1.4 \cdot 10^6$. Figure 3 generalizes the results of investigation of the heat transfer in the separated region: for h = 22 mm the points 1a, 2a, and 3a are for w_1 = 27, 13.7, and 6.75 m/sec; for h = 60 mm the points 1b, 2b, and 3b are for w_1 = = 28.4, 19.6, and 7.4 m/sec; for h = 42 mm the points 1c, 2c, and 3c are for w_1 = 27, 15.6, 11.9, and 5.8 m/sec. All the points obtained for this region were predicted satisfactorily by the relationship (solid line in Fig. 3)

$$N_x = 0.0073\, R_x^{0.8}\, (x/h)^{0.75},$$

$$N_x = \frac{\alpha x}{\lambda}, \qquad R_x = \frac{w_0 x}{\nu},$$

$$w_0 = \frac{G_0}{B(H-h)\gamma_0}.$$

Here N is the Nusselt number, w_0 is the velocity in the initial cross section of the working region; ν is the coefficient of kinematic viscosity and λ the thermal conductivity of the air at the temperature of the flow; x is the distance measured from the base of the plate; α is the heat transfer coefficient (local). In the second region the relationship was simplest and within the limits of experimental accuracy was

$$\frac{N_h}{R_h^{0.8}} = \text{const} = 0.024 \qquad \left(N_h = \frac{\alpha h}{\lambda}, \quad R_h = \frac{w_0 h}{\nu}\right).$$

The region behind the reattachment point was also fairly thoroughly investigated. The range of Reynolds numbers $R_{\Delta x}$ in this region in different experiments was $1.2 \cdot 10^4$ to $1.4 \cdot 10^6$. As the characteristic velocity behind the reattachment point we took the velocity w_1 and from this velocity we calculated the Reynolds number. The experimental points with a slight scatter satisfy Eq. (3) (Fig. 4, where the letters used for the experimental points are the same as Fig. 3)

$$N_{\Delta x} = 0.032 R_{\Delta x}^{0.8} \qquad \left(N_{\Delta x} = \frac{\alpha \Delta x}{\lambda}\right). \tag{3}$$

Here Δx is the distance measured downstream from the point x = = 11 h. The value of the Nusselt number, defined by Eq. (3), was approximately 20% greater than the corresponding value for a smooth plate with $R_x = R_{\Delta x}$.

If $R_{\Delta x}$ in formula (3) is calculated from the velocity in the core of the flow (which of course, is greater than w_1) relationship (3) will be even closer to relationship (1). Thus, we can state that behind the reattachment point the flow will rapidly be reconstructed and will become similar to that in the turbulent boundary layer on a plate beginning at the reattachment point.

In the region of reverse flow, as formula (2) shows, the boundary layer is also turbulent.

The results of the present investigation are similar to those of Seban's investigation [4] of heat transfer in the turbulent separated flow behind a step in a plate surface. This indicates the great similarity of these two forms of separated flows.

REFERENCES

1. R. A. Seban, "Heat transfer to the turbulent separated flow of air downstream of a step in the surface of a plate," Trans. ASME, ser. C, vol. 86, no. 2, 1964.

2. A. Charwat, C. Dewey, J. Roos, and J. Hitz, "An investigation of separated flows, part II, Flow in the cavity and heat transfer," J. Aerospace Sci., vol. 28, no. 7, 1961.

3. E. P. Volchkov and V. Ya. Levchenko, "Effectiveness of a gas screen on a tube surface," PMTF [Journal of Applied Mechanics and Technical Physics], no. 1, 1966.

4. R. A. Seban, "Heat transfer and effectiveness for a turbulent boundary layer with tangential fluid injection," Trans. ASME, ser. C, vol. 82, no. 4, 1960.

5. D. E. Abbot and S. J. Kline, "Experimental investigation of subsonic turbulent flow over single and double backward facing steps," Trans. ASME, ser. D, vol. 84, no. 3, 1962.

6. P. W. Bearman, "Investigation of the flow behind a two-dimensional model with a blunt trailing edge and fitted with splitter plates," J. Fluid Mech. vol. 21, part, 2, 1965.

21 December 1965 Novosibirsk